Observations on Dynamic Qualification Testing of a Component with Nonlinear Deadband Interfaces

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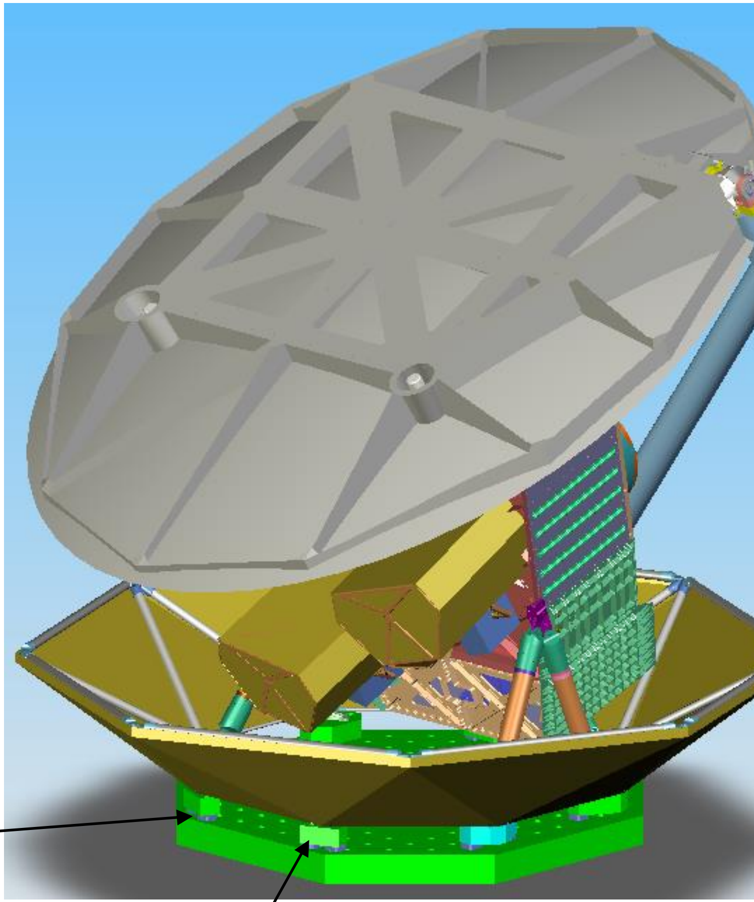
Introduction

- AQUARIUS instrument, one of the NASA's missions managed by Jet Propulsion Laboratory's (JPL), underwent random vibration and acoustic qualification tests
- The instrument was designed to interface with the spacecraft using a series of bipods with mono ball joints and clevises
- The underwent random vibration tests
- As the input to the instrument at the bipod interfaces was increased excessive chatters were observed
- The real-time test data analyses showed strong structural nonlinearity observed due to mono balls clearances and deadbands.
- Higher than expected sigmas attributed to deadbands and gapping of the ball joins and clevises were observed and led us to believe that there are structural workmanship issues related to mono balls with faulty gap tolerances that led to unusual structural nonlinear response behaviour
- After the mono ball and clevis re-work the instrument underwent random vibration test
- Gap in the ball and clevis joints provided the classical and predictable nonlinear structural dynamics behaviour
- In this paper we discuss some observations mad eon the nonlinear behaviour of the structure

AQUARIUS Instrument

- Test Hardware
  - All Flight
  - Total mass 322.5 kg
- Aquarius was not electrically powered during random vibration tests
- Test Fixture and Setup
  - Test fixture plate and 8 fixture blocks
  - Fixture blocks simulate attachment to spacecraft
  - 22 Kistler 9067 force transducers installed in between test fixture blocks and test fixture plate. Force transducer signals will be summed to obtain total force for each of three axes as well as moments.

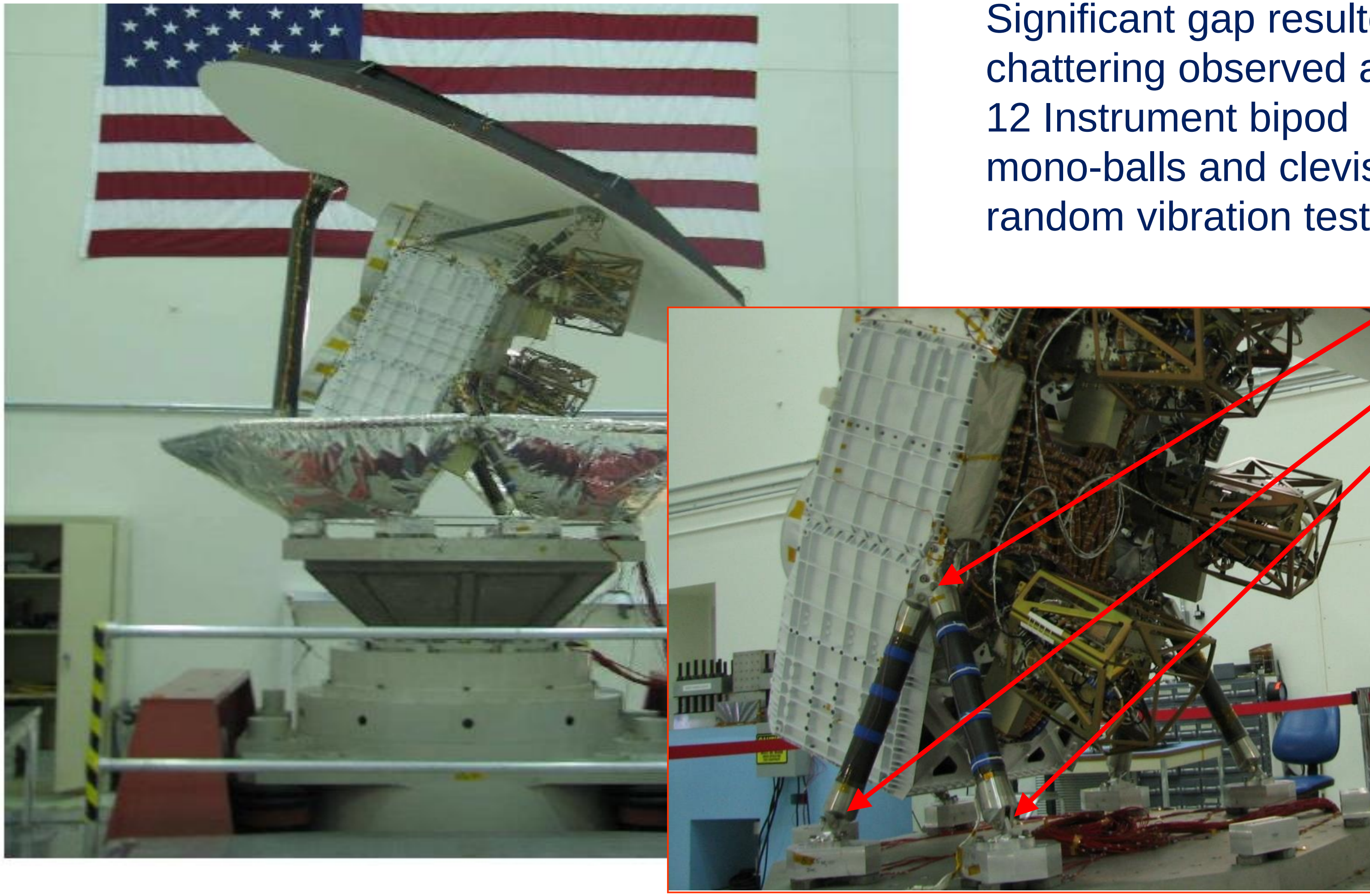


Instrument RV Requirements

Axis	Frequency, Hz	Protoflight Level
X, Y	10 10 – 20 20 – 200 200 – 400 400 Overall	0.0125 g^2 / Hz + 6 dB / Oct. 0.05 g^2 / Hz - 6 dB / Oct. 0.0125 g^2 / Hz 3.78 g_{rms}
Z	10 10 – 20 20 – 200 200 – 400 400 Overall	0.00156g^2 / Hz + 6 dB / Oct. 0.00625 g^2 / Hz - 6 dB / Oct. 0.00156 g^2 / Hz 1.34 g_{rms}

Protoflight (PF) random vibration test in three orthogonal axes for 60 seconds

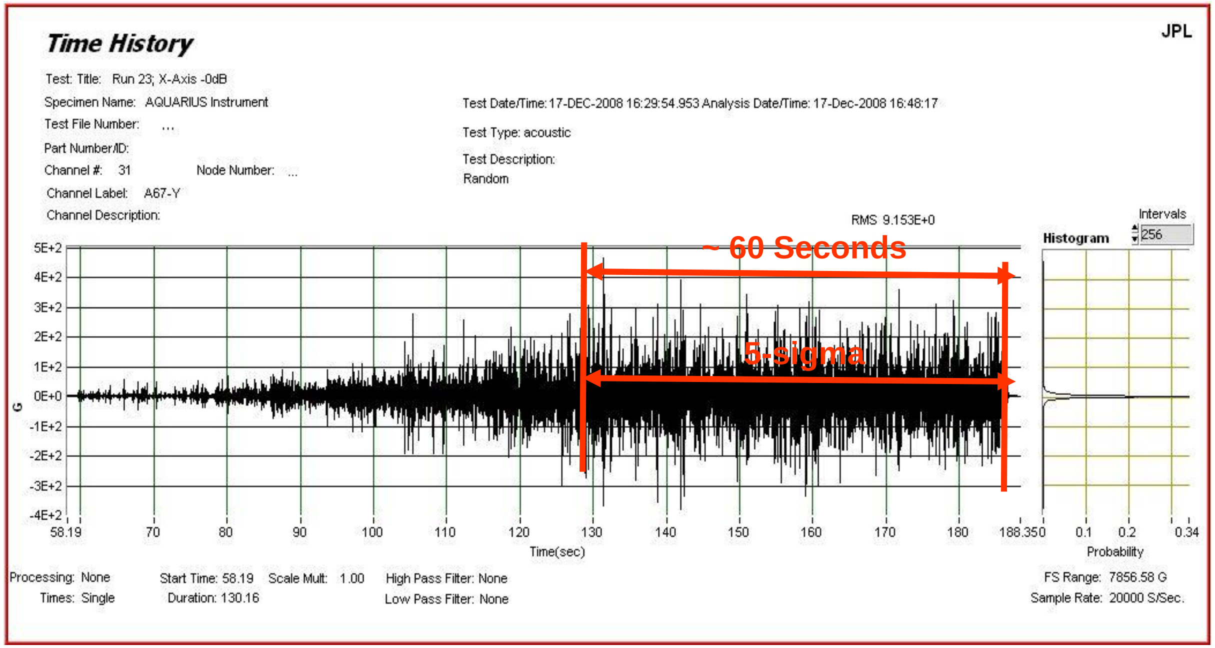
Instrument in Vertical Shaker Axis Configuration



Significant gap resulted in chattering observed at all 12 Instrument bipod mono-balls and clevises random vibration testing

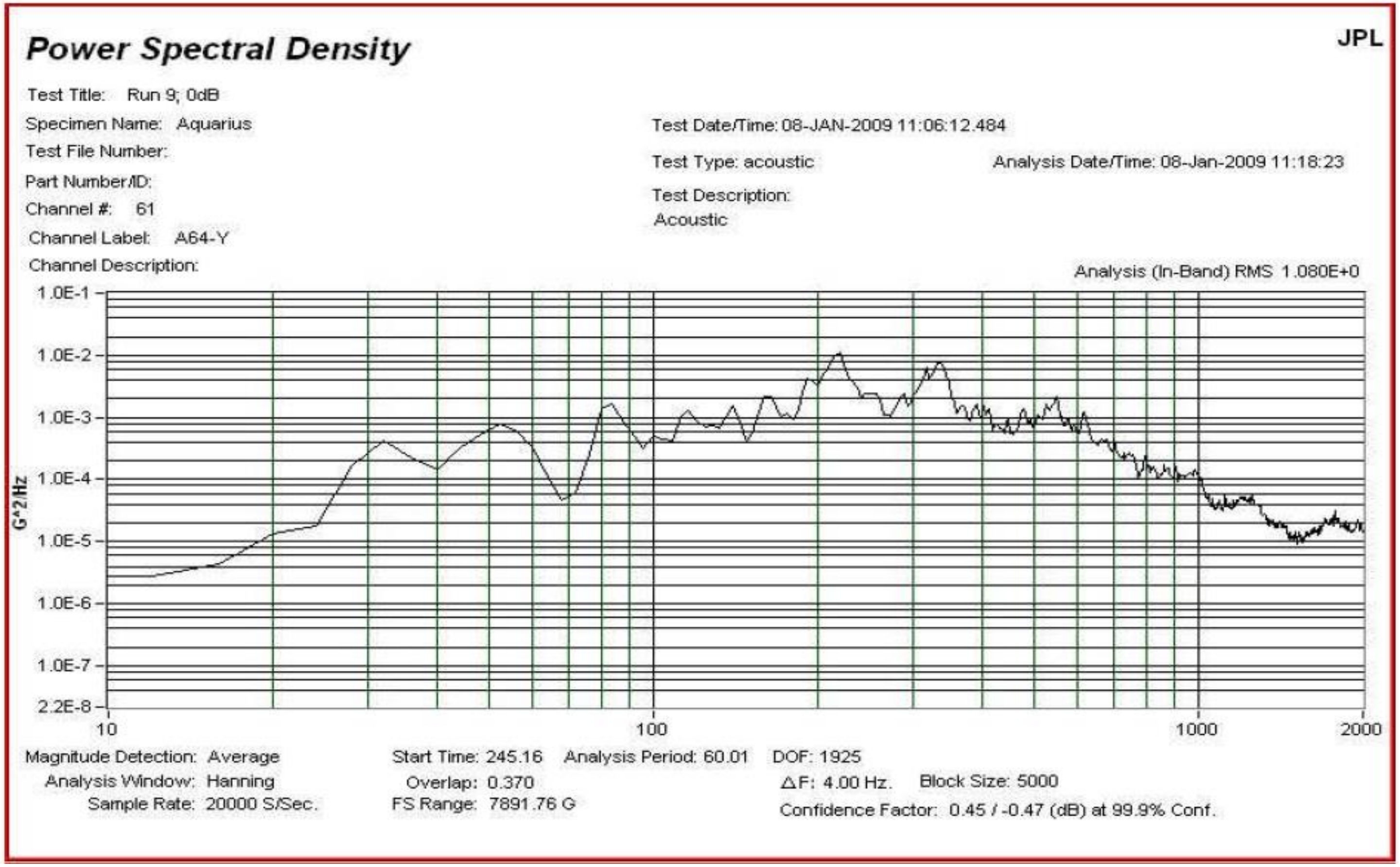
Acceleration Responses Near Bipods

- *Acceleration time history measured near one of the mono-balls. The acceleration rms for full level random vibration test is estimated to be 8.9 where many extreme peaks above 5 sigma had occurred due to the deadband chatters (peak is 450+ g's)*



52 sigma (peak/rms) was observed at monoball joints

Instrument Acoustic Test



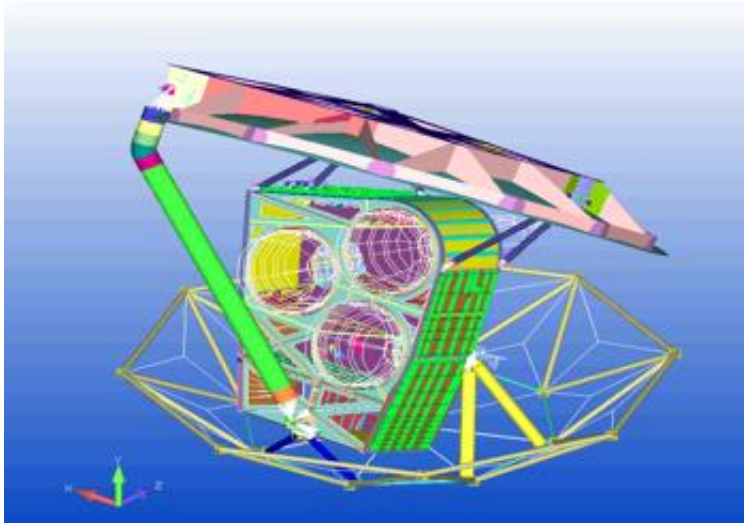
Acceleration PSD measured near one of the bipods. The deadband induced nonlinearity is not as prevalent in acoustic induced vibration as the acoustic energy is low below 100 Hz and it is not effective in displacement of the instrument at its interfaces.

Faulty Mono-balls (A workmanship Issue)

- The extremely nonlinear structural behaviour attributed to bipod interfaces (mono balls and clevises)
- After examination of the joints it was discovered that mono balls had faulty gap tolerances that led to unusual structural nonlinear response behaviour
  - As-installed mono balls, chipping of the liner edges, installation and ball-to-liner tolerance, and potential for mono-ball-to-clevis gapping were discovered
  - Physical evidence of the interfaces also suggested that some of the joints were looser than others, which points to the flaws in workmanship.
- Mono-balls were re-worked and RV penalty test was performed

Pretest FEM Modal Analysis

Mode	Frequency (Hz)	Effective Mass/Inertia						Description
		TX	TY	TZ	RX	RY	RZ	
1	27	1.87%	2.00%	2.03%	4.44%	1.60%	3.00%	1st Reflector Subsystem Mode
2	33	4.25%	1.96%	11.64%	16.83%	20.54%	8.28%	1st Instrument Lateral (Z)/ Bending Mode
3	38	1.92%	4.66%	6.91%	11.13%	0.58%	3.61%	Instrument Bending/Feed/Reflector System
8	41	2.28%	0.25%	2.52%	3.77%	1.01%	2.68%	1st Feed Subassembly Mode
11	44	7.80%	0.19%	0.00%	0.01%	3.39%	6.95%	2nd Feed Subassembly Mode
12	45	4.25%	0.16%	0.24%	0.39%	0.42%	4.95%	3rd Feed Subasssembly Mode
15	50	1.50%	0.59%	5.14%	5.94%	3.16%	2.45%	
16	58	25.57%	0.26%	2.55%	2.45%	0.73%	25.53%	2nd Instrument Lateral (X)/ Feed Horn Mode
17	61	3.98%	0.33%	1.62%	1.25%	2.45%	2.84%	
25	66	0.70%	4.94%	0.85%	0.81%	0.54%	0.78%	
27	67	2.84%	0.03%	0.07%	0.04%	0.55%	2.16%	
30	71	0.81%	4.57%	2.03%	1.83%	0.08%	0.86%	
33	74	3.98%	0.38%	0.18%	0.20%	0.02%	3.81%	
36	79	8.25%	0.02%	0.16%	0.20%	0.19%	8.48%	
37	85	0.01%	6.78%	0.68%	0.71%	1.95%	0.12%	
44	94	1.18%	2.58%	3.99%	3.36%	0.20%	1.07%	
52	102	0.23%	3.01%	8.30%	5.89%	2.21%	0.04%	
75	122	0.06%	3.44%	6.45%	3.86%	0.14%	0.05%	
85	129	0.70%	0.24%	0.10%	0.06%	3.22%	0.54%	
93	138	0.03%	4.32%	3.19%	2.88%	0.18%	0.01%	
366	310	0.00%	0.00%	0.00%	0.00%	4.73%	0.00%	Sunshade Torsional Mode

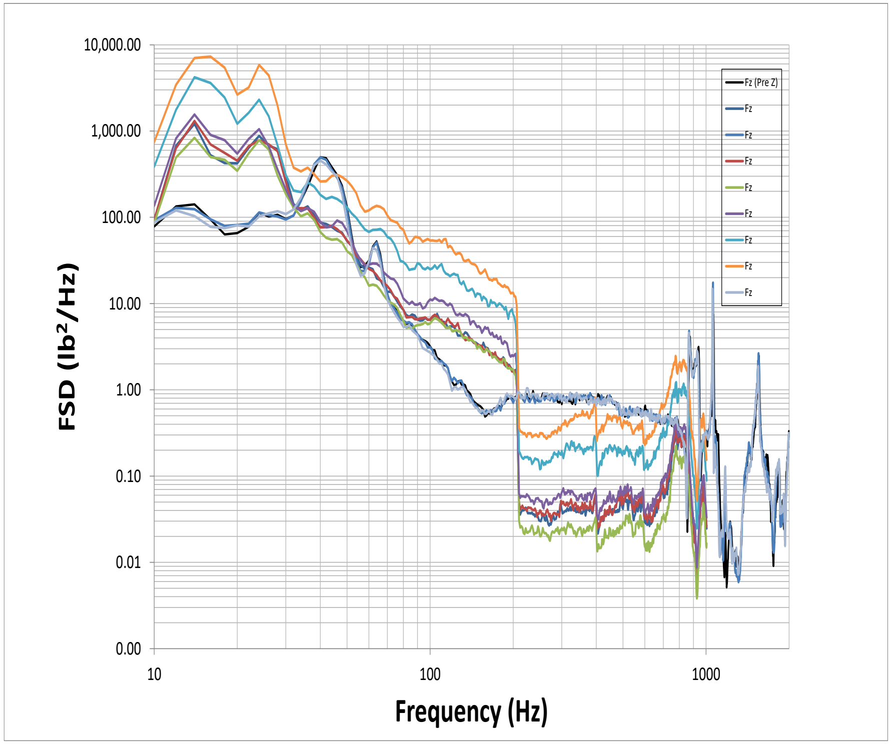


- 90% of Lateral axis effective mass is below 210 Hz
- 90% of Vertical axis effective mass is below 275 Hz

- Typical pretest analysis involves the construction of a linear FEM and the execution of modal analyses (Example of a mode shown here)
- Although this structure is highly nonlinear due deadbands, linear modal analyses with (1) all interfaces constrained and (2) all interfaces free may shed some light into the bounding modal states relative to test levels. A
- Rigorous pretest analysis that is of high value to the testing must involve the modelling of the deadband nonlinearities and time-domain nonlinear simulations

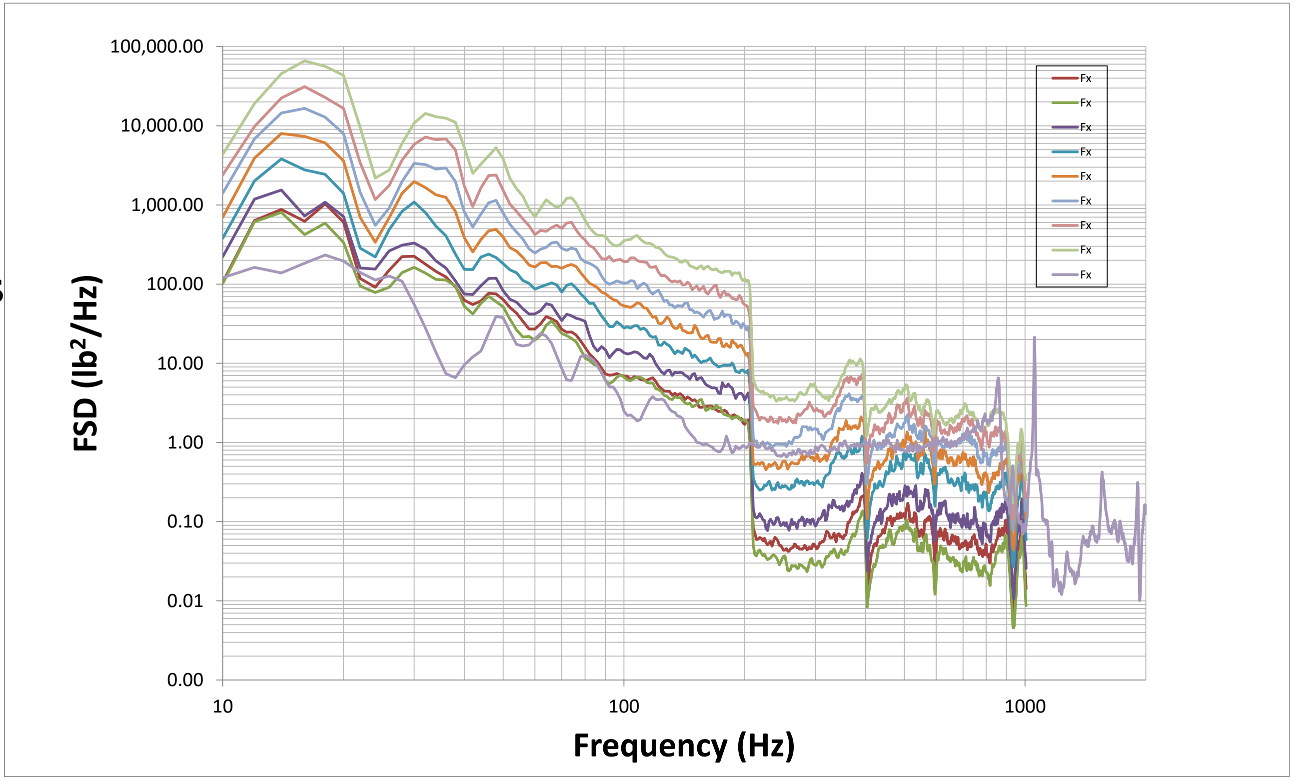
Summed Force PSDs (One of the Lateral Axes; Z-axis)

- spectral shapes computed
  - low input levels (white-noise with 0.45 grms)
  - higher inputs with a 3 dB increment starting from 18 dB below the requirements.
- The following observations are made
  - First, the pre- and post-full level PSD overlays for Z-axis indicates that the primary structural mode of ~40 Hz did not change after the hardware underwent full level random vibration excitation
  - With increasing input to the hardware increased the force spectral shape changed
  - These are the product of the nonlinear system behaviour due to deadbands
  - Further increase in input levels did not cause further change in spectral characteristics



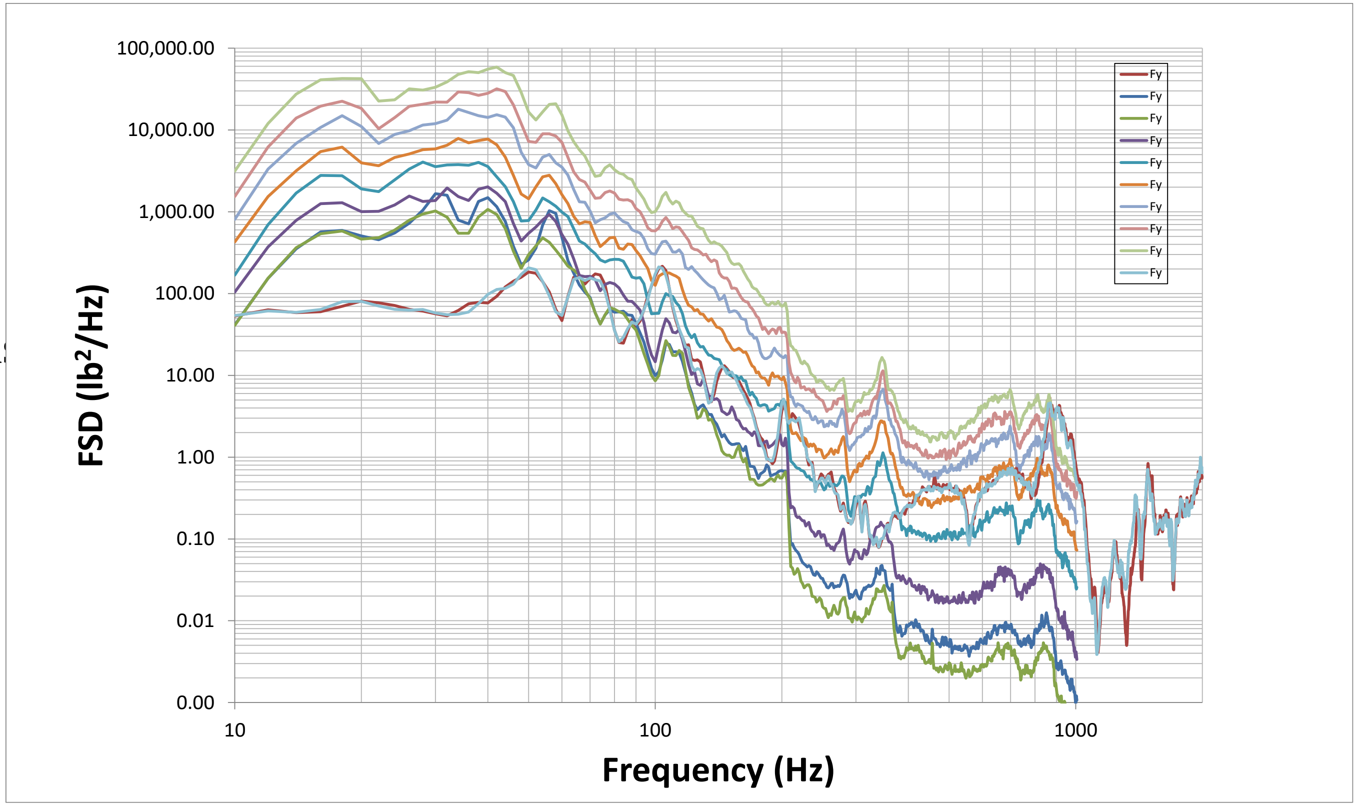
Summed Force PSDs (One of the Lateral Axes; X-axis)

- The same observations as the previous case

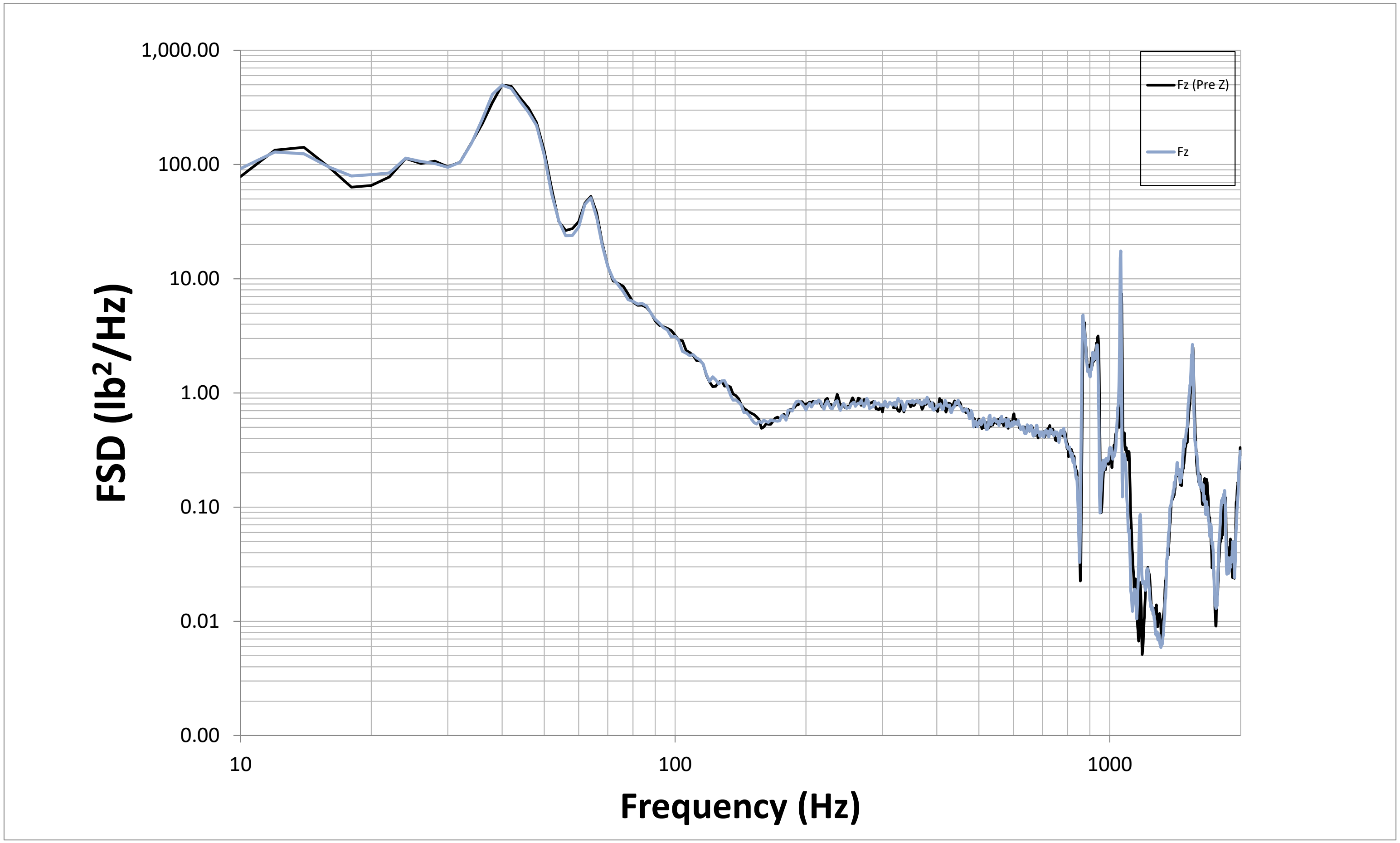


Summed Force PSDs (Vertical Y-axis)

- The same observation[s] ... the previous case

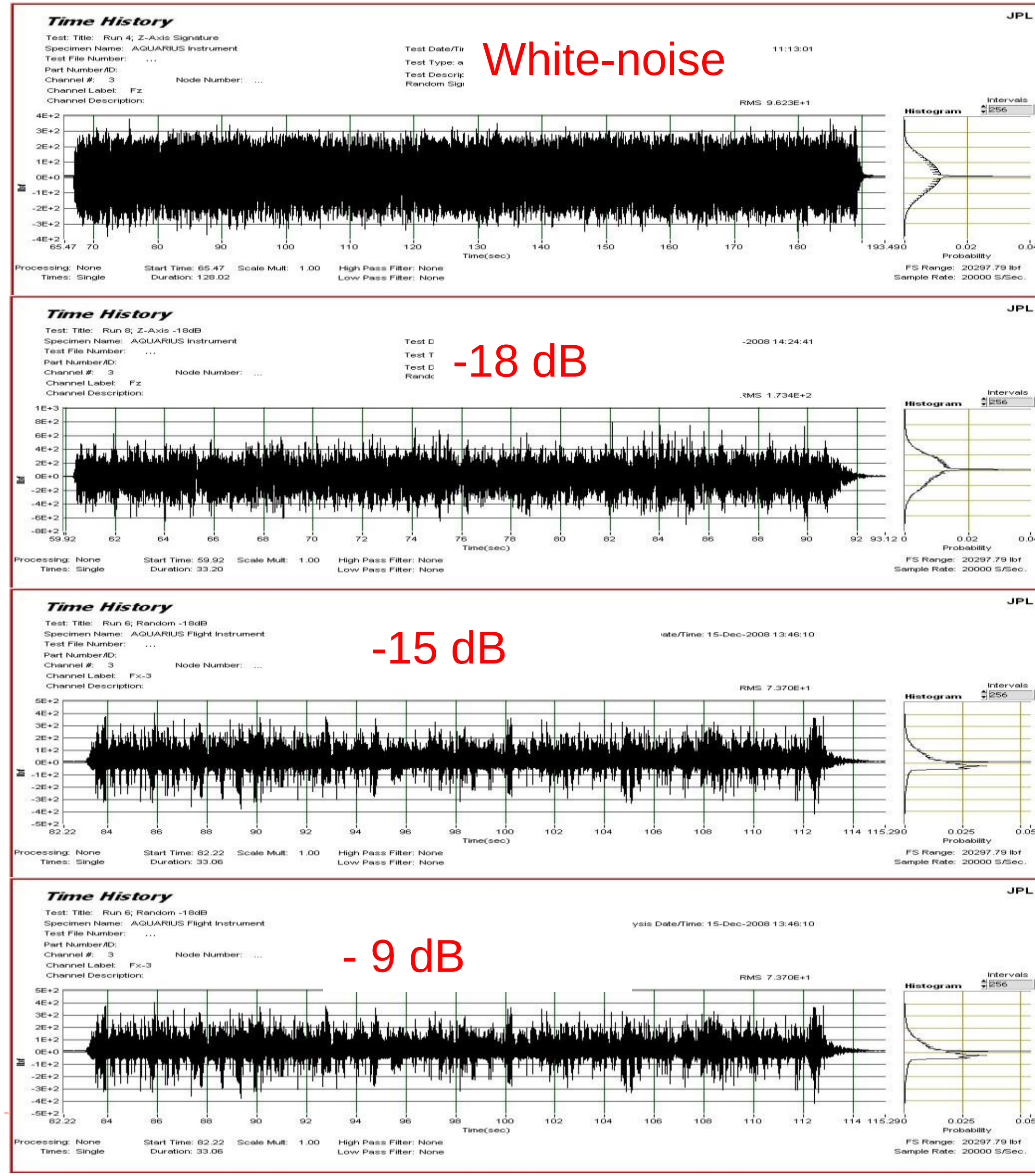


Pre/Post-full Level Overlays



Summed Force Time Histories

- A series of time histories of the interface forces lateral direction (Z-axis) are shown.
- The random vibration responses The departure from the normal distribution of the random responses indicates the impact of the gap is already being felt at the mono ball interfaces.
- More chatter, non-Gaussian distribution indicate impact of the deadband
- The increase in number of chatter and in extreme peaks for these plots qualitatively indicate the displacements of the structures within the mono ball gaps are occurring more frequently (i.e. with faster speed).
- The transition of the slow to fast movement within the gap may have caused the modes to stop shifting



Deadband Behaviour 1/3

- Assume each deadband has a displacement limit [-d, +d]. Each deadband possess 3-states
  - bottomed out at the -d and reacting a positive force,
  - bottomed out at the +d and reacting a negative force, and
  - in transitioning between the two limits and reacting zero force (assuming a pure deadband with no stick/slip friction).
- To demonstrate the complexity of such a nonlinear system, assume the component is supported at 4 interfaces 81 possible modal states – a complex nonlinear system.
- However, some simple reasoning, backed by both nonlinear simulations and test, can be used to explain the behaviour of systems inclusive of deadbands relative to test levels

Deadband Behaviour 2/3

- In a low level test, with “low” defined relative to the deadband limits, the interfaces are transitioning relatively slower between the two limits, therefore, the amount of time spent at zero interface forces becomes longer. With this, the component behaves as if the boundary conditions were free (non-force reacting).
- At higher test levels, again with “higher” defined relative to the deadband limits, the interfaces will transition faster and therefore the amount of time spent in transition (i.e., zero force state) becomes shorter. In this scenario, the component behaves more “linear” with force reacting boundary conditions. In addition, it follows from the same reasoning that any further increase in test levels would not modify this linear behaviour of the deadband nonlinearities.

Deadband Behaviour 3/3

- To quantify the effect of test level on natural frequency, consider a cantilever beam supported at a deadband interface
  - The cantilever's fundamental bending mode will resemble the bending mode of a free-free beam.
  - At higher test levels, the same mode will more closely adhere to the fundamental cantilevered bending mode.
- The fourth order PDE eigenvalue provides the fundamental bending frequency of a free-free beam to be roughly a factor of 6 higher than the same beam cantilevered. Therefore, there is a drop in frequency associated with increase in test levels up to a fully linear behaviour at which the frequency would plateau.
- A drop in primary modal natural frequency with increased test levels stabilizing at the higher test levels.

Summary

- As seen in the AQUARIUS instrument dynamic qualification tests, deadbands can have a significant influence on increasing structural response and changing modal/spectral characteristics.
- In the instrument test, the fundamental frequency of the test article dropped from 40 to 16 Hz with increasing test levels.
- Once the test level was "high enough" (relative to deadband limits), the fundamental frequency "stabilized" at 16 Hz with no further changes in modal/spectral characteristics.
- This is consistent with the expected deadband behaviour and nonlinear simulation findings.
- A rigorous mathematical model is being developed to account for observations made from AQUARIUS Instrument RV test
- The linear FEM analysis lacks accuracy in identifying the instrument primary modes to satisfy flight frequency and loads requirements.
- It is recommended that the pretest analysis for components involving deadband interfaces include time-domain nonlinear simulations.